The measurement of supply air volumes and velocities in cleanrooms Part 1: Supply air volumes

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Abstract

Air supply volumes in cleanrooms are monitored by airflow measuring hoods but these can be inaccurate if used incorrectly. A typical airflow measuring hood was studied and it was found to measure accurately when the air passed evenly out of the hood, as occurred when it was used to measure the volume from terminals with no diffuser and with a four-way diffuser. However, when it was used to measure the air from a swirl diffuser the measurement was 55% greater than the true volume. This inaccuracy was caused by the air swirling round the hood so that at the exit plane, where the measuring matrix was located, the velocity round the outside was very much greater than in the centre. As twelve of the sixteen measuring points were positioned in this outside area, the volume measured was much higher than the actual volume. It was found that this problem could be corrected by partitions in the hood to prevent swirling, or by a permeable membrane at the entrance to the hood to even out the airflow. The back pressure caused by such correction devices was shown to be small and unlikely to influence the air volume measurements. It is suggested that such correction devices should be used in airflow measuring hoods to prevent inaccurate readings caused by swirling or uneven flow from diffusers.

The measurement of supply air velocities, the use of anemometers and the inaccuracies associated with their use will be discussed in Part 2 of this article in the next issue of the journal.

Introduction

Cleanrooms minimise the contamination of products made in manufacturing industries, as well as bacterial infection of patients in hospitals. The most common design of cleanroom utilises 'non-unidirectional airflow'. The ventilation system is similar to that found in hotels, offices etc. but the final air filters, which are of the high efficiency (HEPA and ULPA) type are in the cleanroom ceiling itself, at the terminal end of the supply air ducting. The air supply volumes are much higher than those in normal rooms, being in the range of approximately 20 to 100 air changes per hour. The second type of cleanroom utilises 'unidirectional airflow'. Particle-free air is supplied from a complete HEPA filter ceiling and moves down through the room in a piston-like manner, at a velocity of about 0.45 m/s, before exiting through a perforated floor. A fuller description of cleanrooms, and how they are tested and operated is given by Whyte (2010)[i].

The particulate air cleanliness is directly related to the supply air volume in non-unidirectional airflow cleanrooms and to the supply air velocity, i.e. downflow velocity, in unidirectional airflow cleanrooms (Whyte 2010). These parameters should be monitored throughout the life of the cleanroom, and monitoring intervals are suggested in ISO 14644-2 (2000)[ii].

The air volume supplied to a cleanroom can be accurately measured in the supply air ducts using a Pitot-static tube but this method is normally only used during initial commissioning and any subsequent rebalancing. Routine monitoring is usually carried out using either an airflow measuring hood as shown in Figure 1 or an anemometer. The reasons for using these methods are the extra time required to obtain Pitot-static readings in the air ducts, poor access to the air ducts in most large air conditioning systems, and the greater expertise of the engineers who commission and balance cleanroom ventilation systems, compared to that of those who routinely monitor them.

An airflow measuring hood gathers the air supplied through the ceiling terminal and uses the Pitot-static principle to measure velocity of the air as it exits the hood. A typical arrangement, as shown in Figure 2, has a grid of tubes in the plane where the air exits the hood, with 16 holes positioned to measure and average out the total pressure and 16 holes to measure and average out the static pressure of the air. The difference between the average total pressure and the average static pressure is the dynamic pressure from which the velocity is calculated. Knowing the exit area of the hood, the air volume can be determined.

It has been reported that the measurement methods used for air supply volumes in a cleanroom may give inaccurate readings (Anonymous, 2006)[iii]. As this parameter is of prime importance in determining and maintaining the cleanliness of non-unidirectional airflow cleanrooms, the reasons were investigated.

Figure 1. Airflow measuring hood used on a diffuser in a cleanroom

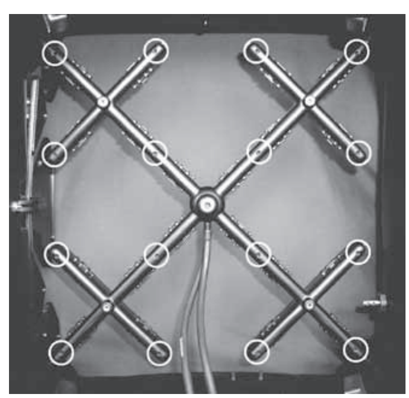

Figure 2. Measuring grid at the exit of an airflow measuring hood with the measuring holes for total pressure circled

Description of test rig and instruments

A simple test rig was set up, incorporating a variable speed fan-filter unit, to allow different diffuser types to be tested, with a low velocity wind tunnel to allow accurate measurements of the volumes required for the tests.

The downstream side of the fan-filter unit had a high efficiency particulate air (HEPA) filter with an 'active' filter face area of 0.54 m x 0.54 m, i.e. 0.29 m^2. (The 'active' area of a high efficiency filter is considered to be the area of the filter media where the air passes through, and does not include the frame). The air filter was protected by a grille. Different types of diffusers could be fitted to this for testing.

A 5 metre low-velocity wind tunnel, of clear plastic sheet, with an air-straightening membrane at its intake, was connected to the upstream side of the fan-filter unit. The tunnel was used to set the air supply volume required for the tests. To do this, a puff of smoke was introduced at the intake to the tunnel and timed with a stop watch as it moved a fixed distance along the tunnel. This enabled the air velocity, and hence the air volume, to be calculated. By varying the fan speed, the exact air supply volume, or filter face velocity, could be obtained to the value required.

The measuring hood used in the experiments was a TSI Model 8375.

Air diffuser types

The supply air to a non-unidirectional airflow cleanroom is usually from a HEPA filter in the ceiling. The filter may have no diffuser, or a diffuser may be fitted. Investigations were carried out with no diffuser and with two diffusers, each of a type commonly used in cleanrooms. These were:

1. A 4-way diffuser of the type shown in Figure 3. This was a 600 mm square Trox Technik Type FD. It throws the air sideways, in four directions, this air entraining and mixing with the cleanroom air.
2. A swirl diffuser of the type shown in Figure 4. This was a 600 mm square Trox Technik Type DLQ. This type of diffuser twists the supply air and mixes it with cleanroom air.

Computational fluid dynamics (CFD)

The design of the diffuser and the angles at which the diffuser vanes are set influence the airflow in the hood. Therefore computational fluid dynamics (CFD) was used to obtain visual representations of this airflow for the respective diffusers and to confirm the experimental results.

Figure 5 shows air streams coming from a HEPA filter with no diffuser fitted passing through a measuring hood. It may be seen that the air flows evenly from the filter to the exit of the hood.

Figure 6 shows airstreams from a 4-way diffuser. The air exits from the diffuser and flows to the outside of the hood where there is some vortexing. There was also some vortexing in front

Figure 3. 4-way air diffuser

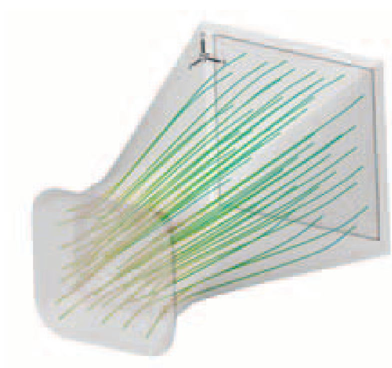

Figure 5. Airflow through a hood coming from a HEPA filter with no diffuser

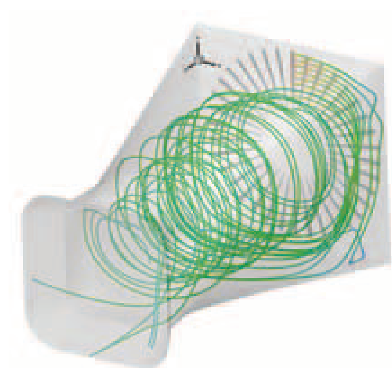

Figure 7. Airflow through a hood coming from a swirl diffuser

Figure 4. Swirl air diffuser. Ribbons or streamers show the direction of the exiting airflow

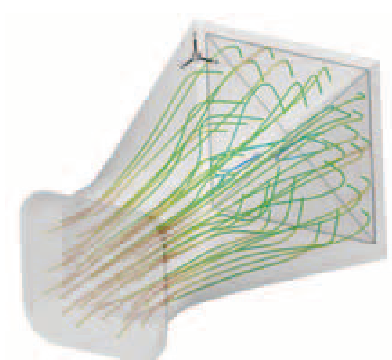

Figure 6. Airflow through a hood coming from a 4-way diffuser

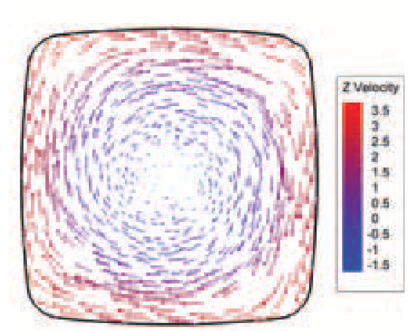

Figure 8. Airflow exiting a hood supplied by a swirl diffuser

of the solid square surface at the centre of the diffuser. It had been anticipated that the air velocity at the exit of the hood would have been uneven but, as seen in Figure 6, this is not so.

Figure 7 shows the airflow in a measuring hood when the air is supplied through a swirl diffuser. It can be seen that the air 'swirls' round the hood. A front view of the exit of the hood is given in Figure 8. As the hood's measuring grid measures velocity in the Z-axis (i.e. in the direction of the airflow from the filter), the magnitude of the velocities in Figure 8 is given in the direction of the Z-axis. The redder the colour and the longer the lines, the greater is the velocity. A high velocity is seen round the outside.

A comparison was made between the velocities at the exit of the swirl diffuser, as found from experimental measurements and as calculated by CFD modelling and a close correlation was found. This suggested that the airflows shown in Figures 5 to 8 are likely to give a good representation of the true airflows.

Experimental measurements of airflows

First of all, tests were carried out to verify the accuracy of the volumes measured by the hood. The air volumes measured by the tunnel were compared with those measured by the hood in order to calibrate the hood. The differences between the two volume flow rates were no greater than 5% at three different nominal velocities.

The air volumes when using the three different diffuser arrangements were then measured by the hood. Table 1 shows that the hood readings and true readings from no-diffuser and a 4-way diffuser differed by -3.9% and -1.0%, respectively. However, the hood readings obtained from the swirl diffuser were 56% higher than the true air volume. This was then investigated further. First of all the velocity was measured at the hood exit using a thermal anemometer and it was found that when no-diffuser, or a 4-way diffuser was used, the velocity was fairly uniform across the exit. However, when a swirl diffuser was used, much of the air exited round the outside and at the centre it moved backwards. This was as predicted by CFD modelling. A better, more accurate experimental method was then devised using the 16 point measuring grid in the hood. Firstly, measurements were taken using all 16 holes. Next, the twelve holes in the outer area of the measuring grid were closed and measurements taken using only the four points in the centre. Finally, the twelve outer holes were reopened, the four centre points were closed and measurements taken from the twelve points in the outer area. Ten tests were carried out for each condition, and the whole procedure repeated five times. The results were averaged out and are shown in Table 2.

The air volumes measured by the hood, when supplied from the swirl diffuser using the 16 normal measuring points, were 55% higher than the readings obtained from both the no-diffuser and the 4-way diffuser. These results are almost identical to those shown in Table 1. The reason for this increased reading is that the outer 12 points of the swirl diffuser picked up substantially higher velocities, which, when averaged with the negative velocities from the inner 4 points, gave a much higher average velocity than for the no-diffuser or the 4-way diffuser.

Devices for airflow uniformity

From the results shown, it is clear that when the air velocity at the hood exit is relatively even across the measuring grid, the air volume measurements are reasonably accurate. This was the situation with no-diffuser and a 4-way diffuser, but not with the swirl diffuser. However, the uneven flow with the swirl diffuser, and other types of diffusers that cause uneven airflow, can be made sufficiently even by either straightening the airflow or making it more uniform.

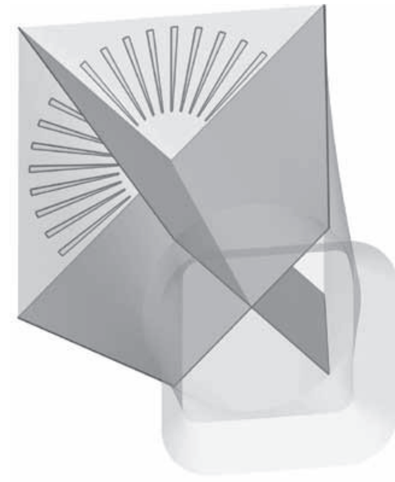

Figure 9. Straightening device – two 'partitions' at 90º

Two methods of straightening the airflow were tested using a) a single partition to split the airflow in the hood and b) two partitions at 90º, as shown in Figure 9. These partitions can be made of the same impervious fabric as the hood and can be sewn into it to allow easy packing and transportation.

To make the airflow more uniform, a permeable membrane was stretched across the intake plane of the hood. The idea was to give sufficient pressure drop to even out the airflow but not too much so as to cause a reduction in airflow. Two materials were investigated, both light-weight and similar to mosquito netting. One was known as 'fine' mesh and the other as 'finest' mesh.

Table 3 shows the results of tests with a swirl diffuser using each of the methods as well a combination of the double partition and the finest membrane. It may be seen that both partition devices gave a noticeable benefit, with the best result coming from the combination.

Table 1. Difference between true and measured volumes with respect to diffuser type

Diffuser type	Difference between true and measured hood volume (%)
None	-3.9
4-way	-1.0
Swirl	+56

Table 2. Volume and velocity measurements at the exit of the hood

	Normal 16 points		Outside 12 points.		Inside 4 points	
	Vol (m^3/hr)	Vel m/s)	Vol (m^3/hr)	Vel (m/s)	Vol (m^3/hr)	Vel (m/s)
No diffuser	476	1.51	476	1.53	504	1.58
4-way diffuser	478	1.53	491	1.55	486	1.55
Swirl diffuser	738	2.23	838	2.69	-118	-0.29

Table 3. Effect of correction devices on the accuracy of the hood measurements when a swirl diffuser was used.

Correction device	Difference between measured and true values (%)
None	56
Fine membrane	16.1
Finest membrane	6.8
Single partition	5.0
Double partition	-3.7
Double partition plus finest membrane	2.4

Some further tests were carried out to verify that none of the devices caused a pressure drop that might affect the volume flow rate. The pressure drops caused by the fine, and the finest membranes were 1.1 Pa and 2.2 Pa respectively. These are a very small percentage of the total losses in an air conditioning plant and associated ductwork used in cleanrooms, where a pressure drop of several hundred Pascals is common, and the HEPA filters alone would contribute between 100 to 150 Pa.

No measurable pressure drops was found when using single and double partitions, and no drop in air volume was therefore measured. None of the correction devices showed a drop in air volume greater than 5%.

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Graham Green is currently a senior lecturer in engineering design in the School of Engineering at the University of Glasgow. He has a Ph.D. degree (Glasgow University), a Masters degree (Loughborough University), and a B.Sc. degree (CNAA). He is a Chartered Engineer (C.Eng.) and is a member of the Institution of Mechanical Engineers (MIMechE) and a Fellow of the Higher Education Academy (FHEA) in the UK. Prior to entering the academic profession, Dr. Green had accumulated 10 years of experience as a Design Engineer and later as a Product Development Manager. His research, teaching interests and expertise relate to Engineering Design; in particular, concept design evaluation, design for reliability, robust design and rapid design and manufacture.

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ii. ISO 14664-2: (2000). "Cleanrooms and Associated Controlled environments – Part 2: Specifications for testing and monitoring to prove continued compliance with ISO 14644-1". International Organization for Standardization, Geneva, Switzerland.

iii. Anonymous: (2006). "Air supply volume inaccuracies", The Cleanroom Monitor, 55, April, pp 6-7.